DCATT

Responses to Action Items from the Mechanical Systems Peer Review
8/16/98

Document Controlled by:

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Peer Review Action Item Responses

#	Action Item	Engr	Date Closed
1.	Re-look at unbolting and cleaning step at GSFC or at U of Az. [Seery]	JB	10/28/98
2.	What is requirement for the stiffness of the Stewart platform bucket mounting ring [translation: the entire hexapod and bucket assembly]. [Seery]	JB MM DR EY MW	1/22/99
3.	Document your assembly procedure of the hexapods. [Seery]	JB	1/22/99
4.	Assess the impact of the hexapods on the non-kinematically mounted secondary (and primary). [translation: ?] [Seery]	JB MM	1/22/99
5.	Assess the overall stability of the hex ring assembly with secondary and with the 6 aluminum legs. [Seery]	DR MM	12/4/98
6.	Calculate the resonant frequency of the metering structure. [Seery]	DR MM	12/4/98
7.	Call Vukabrotovich [or some other expert] on hindle mount epoxy joint scheme. [Seery]	DR	12/4/98
8.	Take into account the distortions in the optical table between hard mount and floating conditions [Seery]	DR MM	10/2/98
9.	Complete the design of the Zygo alignment tower. [Seery]	DR	In progress
10.	Photo document all the hardware. [Seery]	DR	10/2/98
11.	Complete the analysis of the segment-segment collision avoidance. [Seery]	TC	In progress
12.	Investigate the capability of the screw jacks to resist backdriving under the expected load. A brake may be required. [Niemeyer]	DR	9/14/98
13.	Determine the horizontal stiffness and load capability of the machine screw jacks, isolators, and AC Flat Support Frame interface to the machine screws. [Hakun, Wallace]	DR MM	12/4/98
14.	Determine amount of squeeze of AC flat support O-Rings.	DR	9/14/98
15.	Write a procedure for installation of Collar Assembly onto AC flat.	DR	9/14/98
16.	Write a procedure for receiving AC flat from Zygo, installing collar assembly, rotating, moving into the CIA room. Include an inspection for new dimensions of the AC flat and the shipping method from Zygo to GSFC. [Hakun, Moya]	DR	10/2/98
17.	Do an analysis on what happens if the AC Flat Support Frame is tipped at maximum amount on jack screws. Will there be a safety hazard? [Moya, Hakun]	DR MM	12/4/98
18.	Check ball bearing plate to steel levelling pad interface static friction coefficient. If too high, select more compatible materials [Davila]	DR	10/2/98

19.	Recommend closing off tubes on Upper T-frame to prevent contamination of optics later on from dust buildup. [Wallace]	DR	9/14/98
20.	Consider stress relieving secondary mirror.	JB AM	10/28/98
21.	Determine the stress on the AC flat during 2 point rotation. Do a Weibull analysis [Hakun]	DR MM	10/9/98
22.	Verify safe condition of AC flat during shipment from Zygo, given that it will no longer have same thickness as before. Does the collar still work? [Hakun, Moya, Wallace]	DR	10/2/98
23.	Analyze the self-weight gravity sag of secondary mirror design	JB MM	1/22/99
24.	Determine worst-case shock loads and results on DCATT caused by isolator failure, either abrupt loss of pressure or wild swing back and forth.	DR MM	2/11/99
25.	Determine modes of entire DCATT assembly and its response to the acceleration environment in clean room. Determine motion of AC Flat in hindle mounts. Determine motion of primary and secondary mirrors. Determine if significant loads are induced into epoxy joints or jack screws	DR MM	In progress
26.	To provide more lateral stiffness, reduce the extended length of the 1-Ton machine screw jack. Set nominal position at +5.5 inch/-.5 inch, not +/- 3 inches travel, for example. [Ryan]	DR	9/14/98
27.	Recommend that all welded structures are stress relieved via heat treatment to minimize concerns of creep. [Hakun,Moya, Wallace]	DR	10/9/98
28.	Recommend that Materials Group "bless" design of epoxy joint and the epoxy chosen. [Moya, Hakun]	DR	12/4/98
29.	Recommend revisiting design of Hindle mount mushrooms and use of spherical washers. Explain the function that the spherical washers provide.	DR	9/14/98
30.	What is resolution requirement on tip/tilt of AC flat?	DR	9/14/98
31.	Machine screws may drip oil and contaminate mirror. Recommend bagging screws or building a "well" around screws. The well may be advisable for safety reasons anyway, to ensure if jack screws tipped over during earthquake.	DR	9/14/98
32.	What is the strength of the 1/4-20 levelling pads?	DR	10/9/98
33.	Recommend measuring frame before it goes into the clean room. Understand its out-of-straightness condition before mounting AO Bench. [Niemeyer, et al]	DR	9/14/98
34.	Go over epoxy procedure with Israel Moya.	DR	12/18/98
35.	Recommend getting Quality Assurance involved on the epoxy bond. [Wallace]	DR	12/4/98
36.	Recommend writing up a test plan and executing test for finding the backdrive load on machine screws, the horizontal load capability, and how safety will be assured during operation of machine screws if driven to full tip/tilt [Wallace]	DR	10/9/98

03/22/99

37.	Recommend butt milling support beams [Wallace]	DR	10/2/98
38.	Recommend checking temperature of CIA room at various sites and elevations [Wallace]	DR	10/9/98
39.	Recommend measuring flatness of mounting pads and AO Bench frame. [Wallace]	DR	9/14/98
40.	Write a briefing paper which describes the various metering structures considered and why present design was chosen. Include also preliminary design of Serrier truss metering structure. Include cost/schedule impacts for going with Serrier truss. Include also results of metering structure characterization tests.	DR	12/18/98
41.	Consider hard stops under AO Bench Frame in case of failure mode on isolators which could cause entire tower stackup to sway uncontrollably.	DR	10/9/98
42.	Write hexapod prototype test plan.	JB	1/22/99
43.	What is mass of lightweighted segment compared to un-pocketed segment?	JB	12/4/98
44.	Consider a modal survey of hexapod assembly to verify model.	JB	9/17/98
45.	Can an actuator be removed and replaced if it is broken?	JB	9/17/98

AI #1:
Re-look at unbolting and cleaning step at GSFC or at U of Az. [Seery].

ACTION ITEM IS CLOSED: 10/28/98

Responsible Engr: Jamie Britt

Project Manager: Claudia LeBoeuf

This issue was resolved at a meeting with Pam Davila, Claudia LeBoeuf, Jim Lyons, Jamie Britt, and Dave Robinson on 9/10/98. John Hagopian's inputs were included through off-line discussions with Pam Davila prior to the meeting.

All present at the meeting decided that no unbolting test would take place at Optical Sciences Center (U. of Az.). Jim Lyons' experience suggests that partial unbolting of the mirrors will not give reliable or meaningful information about springing. Furthermore, if springing occurs, it is unlikely that further polishing would be able to correct the problem.

Steps were taken during fabrication to minimize springing in the mirrors. If an unacceptable amount of springing occurs when the mirrors are unbolted from the baseplate after polishing, changes will be made to the hexapod-mirror interface in an attempt to recreate fabrication mounting conditions in the operational system. For this plan, it is most appropriate to unbolt the mirrors and test for springing at GSFC after the bolted mirror is characterized.

AI #2:

What is requirement for the stiffness of the Stewart platform bucket mounting ring [translation: the entire hexapod and bucket assembly]. [Seery]

In Progress:

Responsible Engr: Jamie Britt, Eric Young

Project Manager: Claudia LeBoeuf

The key requirement is motion of a primary mirror segment relative to the primary baseplate. The requirement for stiffness of the hexapod is driven by the need for the line-of-sight (LOS) jitter to be within what the proposed wavefront sensing technique can handle. These numbers are still being worked to nail down an appropriate requirement for the wavefront sensing and to divide it appropriately into jitter budget values for each optical component. The requirement at the wavefront sensing level is still being kept at 1/10 of a CCD camera pixel, though this value may have some margin.

A mechanical analysis of the hexapod with the old bucket design was completed with the hexapods in their operational configuration. The first and second modes were at 117 Hz and were lateral movements of the segment. We are now changing to a system in which the buckets are eliminated and the hexapods sit on top of a newly designed baseplate.   This will greatly improve the ease of changing out an actuator in the case of a failure. The hexapod by itself has a first and second mode at 131 Hz per analysis. The jitter is being calculated as part of an overall analysis of the entire structural assembly using measured input vibration data. The prototype hexapod underwent a low level sine-sweep test the week of January 18 to determine the actual resonant frequencies. This data compared very well to the analytical results and the current model is considered validated.

AI #3
Document your assembly procedure of the hexapods. [Seery]

ACTION ITEM IS CLOSED

Responsible Engr:

Project Manager:

The hexapod assembly procedure is documented in DCATT-MECH-PROC-005.

AI #4

Assess the impact of the hexapods on the non-kinematically mounted secondary (and primary). [Seery]

In progress:

Responsible Engr: Jamie Britt, Eric Young

Project Manager: Claudia LeBoeuf

See response to Action Item # 2.

AI #5:
Assess the overall stability of the hex ring assembly with secondary and with the 6 aluminum legs. [Seery]

ACTION ITEM IS CLOSED: 12/4/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

Three effects cause misalignment between the primary and secondary mirrors: microcreep, vibration and thermal expansion. Microcreep is minimized by using stress relieved (6061-T651) structural members in the metering structure.

To better understand vibration issues, the metering structure with kevlar bracing was tested at the modal facility in building 15 during the week of 9/28/98, and the results were used to fine tune the FEM model of the metering structure and to make some minor modifications to improve the structure. The fine tuned FEM model was then used to analyze possible replacement of the kevlar bracing with either aluminum bar bracing or solid shear panels. The table below shows the results of the testing (for the kevlar) and the analysis (aluminum bracing, shear panels, correlated kevlar models).

For the kevlar and aluminum bracing, the 1st and 2nd modes were in bending in the X and Y axes. The 3rd mode was torsion around the center of the hex ring (the optical axis). For the 0.25 thick shear panels, the 1st mode was in torsion (76 Hz) and the 2nd and 3rd in bending (115 Hz). Of primary importance in all cases are the bending modes. The torsion mode should have negligible effect on the optical performance.

The frequency data was used to correlate the finite element model. After correlation, the model was used to predict the RMS motion of the secondary mirror vertex in microinches. Random vibration PSD data was taken in the CIA room on top of the AO Bench with the vibration isolation system on.   This data was input into the FEM and the secondary displacement amount was determined. These numbers may be conservatively high due to the fact that the measured levels were at or below the noise level of the accelerometers used. Micro-g accelerometers were recently procured and more sensitive measurements taken. An updated analysis using the new data is in progress. The data shown still provides a good comparison between the designs.

Configuration	Relevant Fundamental Frequency (bending)	Displacement of SM relative to PM due to a conservatively high estimate of vibration input from CIA room floor – actuals may be lower
Unbraced legs	8.7 Hz	100 μi
2 strands Kevlar	13.5	50 μi
4 strands Kevlar	16.5	30 μ
10 strands Kevlar	22.3	10 μi
0.25 x 0.5" aluminum bracing	58	3.2 μi
Shear panels 0.125" thick	61 bending 76 torsion	2.1 μi
Shear panels 0.25" thick	76 torsion 115 bending	1.6 μi

The last cause of stability concerns optical misalignments in defocus due to thermal expansion of the metering structure. This factor is minimized by using an all-aluminum telescope. As the metering legs expand with temperature, the radii of the primary and secondary mirrors also change. The overall result is a nearly zero change in telescope performance. The project agreed on this approach in late 1997.

The CIA room was characterized for temperature in the fall of 1997. The average temperature was 66 degrees F +/- 2 degrees. The room never changed more than 3 degrees F in a 24 hour period. Assuming a maximum temperature swing of 3 degrees per day would be conservative as DCATT will not be operated 24 hours per day.

AI #6:
Calculate the resonant frequency of the metering structure. [Seery]

ACTION ITEM IS CLOSED: 12/4/98

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

This action item is being expanded to cover the frequency of the metering structure at various configurations and secondary mirror motion relative to the primary mirror.

The resonant frequency of the metering structure without any straps or bracing is 8.7 hertz, according to testing done at the GSFC modal facility in building 15. The frequency increases with every strand of Kevlar that is looped around the structure. The test results are as follows:

Test #	# Strands	Tension	1st and 2nd Mode	3rd Mode
1	0	0	8.7 Hz	15.2 Hz
2	2	25	13.5 Hz	21.4 Hz
3	2	50	13.5 Hz	21.4 Hz
4	4	60	16.5 Hz	26.8 Hz

The tension in the Kevlar strands had no effect on the frequency response. The number of strands increased the stiffness of the structure.

A finite element model was completed and correlated to the above results. The FEM showed that the displacement of the secondary mirror relative to the primary mirror depended on the bracing used, as shown below:

Configuration	Relevant Fundamental Frequency	Displacement of SM relative to PM due to a conservatively high estimate of vibration input from CIA room floor – actuals may be lower
Unbraced legs	8.7 Hz	100 µI
2 strands Kevlar	13.5	50 µI
4 strands Kevlar	16.5	30 µI
10 strands Kevlar	22.3	10 µI
0.25 x 0.5" aluminum bracing	58	3.2 µI
Shear panels 0.125" thick	61 bending 76 torsion	2.1 µI
Shear panels 0.25" thick	76 torsion 115 bending	1.6 µI

The requirement for the secondary mirror displacement has not yet been fully defined, but it is estimated that it will be < 1 µi. Based on these analysis results, the Kevlar bracing method had to be discarded and an aluminum bracing method substituted. It is still hoped that the aluminum bracing method will be sufficient as opposed to requiring the shear panels, as it is very desirable (though not mandatory) to have visible access to the primary and secondary mirrors during the testing for various optical measurements. The entire DCATT assembly will be characterized both optically and with accelerometers in January, 1999. The aluminum braces will be in place in the metering structure. If there is too much jitter, then a full shear panel design will be implemented on the metering structure.

An alternate metering structure design utilizing a Surrieur truss type structure was also analyzed but was found to be no more efficient stiffness-wise than the current metering structure with aluminum braces, even when mounted to an infinitely stiff base (an actual design base would not be infinitely stiff and would reduce the overall frequency). A report will be out on this shortly (DCATT-MECH-DESI-002)

AI #7

*Call Vukabrotovich [or some other expert] on hindle mount epoxy joint scheme
[Seery]*

ACTION ITEM IS CLOSED: 12/4/98

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

This is a good idea. However due to resource limitations and the limited availability of outside experts to spend time on this issue, no outside experts will be consulted.

Several Goddard experts have been consulted, however, including Michael Viens and Carol Clatterbuck of the Materials Branch, Jim Lyons and Bill Eichorn of the Optics Branch, Armando Morell and Israel Moya of the Electromechanical Branch.

The bonding scheme will be tested on the actual autocollimating flat. The plan is to follow the bonding procedure and then after curing to suspend the flat no more than half an inch for at least one month to test the short and long term stability of the bond. If the bond survives one month, it should be suitable for use on DCATT.

AI #8:
Take into account the distortions in the optical table between hard mount and floating conditions [Seery]

ACTION ITEM IS CLOSED: 10/2/98

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

There are two issues here that affect distortions in the optical systems on the AO Bench: (1) how the AO Bench is mounted to the AO Bench Frame, and (2) whether the AO Bench Frame can induce distortions into optical system when it changes from being simply supported on the floor to floating on pneumatic isolators.

Issue (1)

Two factors can cause optical misalignments, thermal expansion and vibration.

The AO Bench Frame is made of ASTM A500 steel tubing. The 5' x 12' x 18" thick Newport RS-4000 AO Bench is constructed of 400-series stainless steel face sheet 3/16" thick with a steel honeycomb core. The bottom face sheet is steel. The AO Bench will rest on 6 mount plates bolted to the AO Bench Frame. Six angle brackets will be loosely bolted into the underside of the AO Bench and the AO Bench Frame for safety reasons. These angle brackets are not in the load path of the AO Bench.

Because both the AO Bench and the AO Bench frame are steel, there will be no differential thermal expansion. Thus no optical distortions will occur in the AO Bench due to thermal expansion between it and the AO Bench Frame.

Non-uniform thermal changes in the room or local heat sources could cause bending of the RS-4000 table itself. According to Newport, the tables have a thermal time constant of 1 hour or more. The effect of bending was analyzed in 9/97. If the top of the AO Bench is heated 3 degrees F and the bottom surface remains constant, the table will bend (bow) 27.6 arc seconds. A 3-degree F bulk temperature change in the room will expand the table 0.0026 inches measured diagonally from a bottom corner to the top opposite corner.

According to Newport's vendor information, the normal static deflection of the table will be ~7 microinches per pound of mass placed in the center of the table.   Once the optical equipment is mounted on the table and aligned, no further movement of the equipment is induced by this sag.
Vibration will occur no matter if the table is floating or hard mounted to the AO bench frame. Vibration is controlled by (a) locating in a quiet room on a seismic block to minimize input vibrations, (b) putting DCATT on Newport I-2000 pneumatic isolators that attenuate frequencies significantly.

Issue (2)

During initial optical alignment testing, John Hagopian would like to level the AO Bench relative to gravity while the pneumatic isolators are "off". Later the isolators will be turned "on" and the AO Bench Frame will "float" on a cushion of air. The concern is whether the transition from these two conditions could cause deformations on the AO Bench which cause optical misalignment.

The AO Bench Frame will be leveled with 4 hydraulic or machine screw jacks sitting on the floor of the CIA room. They will be placed at the farthest 4 corners of the AO Bench Frame. Thus the load of the DCATT assembly will be transferred through the box beams of the AO Bench Frame and into the 4 jacks and then into the floor. When the isolators are activated, the load of the DCATT assembly will have a new path. The load will be taken out at 6 points at the interface with the isolators.

A small angular re-orientation of the AO Bench relative to gravity by itself will cause some deflection in the table. A one degree rotation may cause as much as 20 microinches of deflection across the table [1700 lbf * sin(1) * 7 μi per lbf]

The AO Bench Frame will deflect approximately 0.05" when the AO Bench is attached to it and the AO Bench Frame is lifted by four jacks. It is in this configuration that the AO Bench will be shimmed to the AO Bench Frame and the angle brackets loosely attached for safety. When the Telescope Support Frame, Telescope, and AC Flat Support Frame are lowered onto the three pads, and additional deflection of approximately 0.05" will occur. Since the AO Bench is not fixed to the AO Bench Frame, the AO Bench Frame is free to sag out from underneath the AO Bench. The AO Bench will not see any forces and the aft optics will not move relative to each other. The AO Bench will then be like a plane resting on a concave surface, with the concavity on the order of 0.05".

It is in this configuration that the isolators will be turned "on". The load path in the AO Bench Frame will change to take the forces out to the isolators. The total sag of the AO Bench Frame will be less than before, since there are now 2 more supports located approximately in the center of the length of the deflected beams. This change in deflection will not induce loads into the AO Bench but will change the relative position of the Telescope to the 45-degree fold flat on the AO Bench no more than 0.1" in despace.

AI #9

Complete the design of the Zygo alignment tower. [Seery]

ACTION ITEM IS CLOSED

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

The Zygo alignment tower is currently in the preliminary stages of design. It is expected to be fabricated in March, 1999.

AI #10

Photo document all the hardware. [Seery]

ACTION ITEM IS CLOSED: 10/2/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

All the hardware will be photographed using the 543 Branch digital camera throughout the buildup of DCATT. Photographs can be downloaded from the DCATT website photo gallery.

http://dcatt.gsfc.nasa.gov/

AI # 11

Complete the analysis of the segment-segment collision avoidance. [Seery]

In Progress:

Responsible Engr: Tim Carnahan

Project Manager: Claudia LeBoeuf

A preliminary assessment of the issue was done by Laura Burns using Matlab to run through a number of cases to determine first of all the scope of the challenge and secondly the best method of dealing with collision avoidance in-situ in the testbed. Her conclusions, presented at a meeting on September 29, 1998, were:

1) Segment-to-segment collisions were possible given the 6 mm gap size and the range of motion of the hexapods.

2) Setting blanket actuator limits would be difficult due to the large variety of possible paths they could take and large variety of positions that neighboring segments could be in.

3) Look up tables would also not be a good solution due to the vast number of possible position interactions.

4) Three other analysis methods were compared, boundary definition, planar intersection, and proximity calculation. Any of the three could be used in-situ with the hardware.

It was decided at the meeting that the planar intersection method would be best, based on efficiency and on the fact that this method couples will with the I-Grip collision detection software that we have at GSFC. Once the system is modeled in I-Grip, the software can be used in-situ to quickly determine whether or not an entire trajectory of point-by-point motions are safe prior to implementation of control commands.

Tim Carnahan, who has extensive experience with collision avoidance analysis and the I-Grip software, was assigned to do this work. He currently has the I-Grip models of the actuated segments finished and will be beginning some simulation work shortly.

03/22/99

AI #12

Investigate the capability of the screw jacks to resist backdriving under the expected load. A brake may be required. [Niemeyer]

ACTION ITEM IS CLOSED: 9/14/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

Backdriving is a common problem in machine screw jacks with low gear ratios, or in all jacks under vibration. The DCATT jacks have a very high gear ratio of 100:1 and will be operated in a vibration-isolated environment so no backdriving was expected. Backdriving can always be solved with a brake.

A test was performed 9/8/98 to determine whether the 1-ton Duff Norton M-2501 machine screw jacks would backdrive. The lifting height of the jack was set at 0.5 inches and a dial indicator was attached to the input drive shaft and touching a flat steel block. The dial indicator measured rotation of the shaft within 0.0005 inches, which translates into better than 0.2 arc-second tip/tilt resolution for the AC Flat Support Frame. The machine screw jack was taken to a load cell in building 7 and a load of 500, 1000, and 1500 pounds was applied. In each case an initial compression set was observed as the load went from 0 - 500 pounds, 500 - 1000 pounds, and 1000 - 1500 pounds. This compression set was 0.002 inches per 500 pounds of load. After a 5-minute hold, no backdriving was observed during a hold of 5 minutes. Since the maximum load expected on each of the three jackscrews on DCATT is 477 pounds, we can conclude that no backdriving will occur. Thus no brake is required on the jackscrew motor.

AI #13:
Determine the horizontal stiffness and load capability of the machine screw jacks, isolators, and AC Flat Support Frame interface to the machine screws. [Hakun, Wallace]

ACTION ITEM IS CLOSED: 12/4/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

The Duff-Norton MS-2501 machine screw jack is rated to lift 2000 pounds vertically. Its actual failure point is at 4000 pounds, according to data supplied by the vendor. Its horizontal load capability (side thrust rating) is 89 pounds for a 400 pound compression load with a 3" raise. The three DCATT jackscrews lift 1352 pounds less than 1" above their lowest position. When they are all at the same height, the load is spread evenly at 1352/3 = 451 pounds apiece.

The jackscrews were tested on 12/3/98 to verify whether they would see a side load that could cause a safety hazard. In the actual AC Flat Support Frame, the jacks were raised 3.75 inches one at a time. In one configuration, two jacks were up 3.75 inches and the other was at its lowest position. In another configuration, one jack was up 3.75 inches and the other two were at their lowest position. In neither of these situations were there any binding, breaking, or other anomalies noted. In this latter configuration, a tilt of 2.88 degrees exists. The sine of 2.88 degrees times the total mass of the AC Flat Support Frame equals 68 pounds of lateral force distributed unevenly on the 3 jackscrews. The test results showed that the jackscrews are safe when the maximum difference between jacks is less than 3.75 inches. There is a conservative distance since the jacks were not tested to failure.

The Newport I-2000 isolators are 13.5 inches tall, with a 1.3" total vertical piston travel. Each isolator is bolted to a .75" thick aluminum isolator plate which is in turn bolted to the AO Bench Frame. For the AO Bench Frame to tip over the isolators, a lateral force would have to be exerted great enough to cause the resulting force vector (originating at the CG) to point out of the rectangular footprint formed by the 6 isolators. The tip-over force has been calculated to be 4377 pounds. This force is lower than the force required to shear the four _-20 screws holding each isolator to the AO Bench Frame. The side force allowable on the isolator itself is not known but each isolator can handle up to 2000 pounds vertical load. Using side load strength of 10% of its vertical strength still provides 1200 pounds of side load available over 6 isolators.

AI #14
Determine amount of squeeze of AC flat support O-Rings

ACTION ITEM IS CLOSED: 9/14/98

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

The O-rings used on the collar assembly are .25" diameter Viton o-rings. They sit in a groove that is .300" wide and .180" deep. This design was based on the non-pressure seal o-ring groove design information provided in the Parker O-ring Handbook. The gap that exists between the top of the aluminum lip segment (part #2022778) and the AC Flat was measured at 0.065" with a feeler gage accurate to 0.001". This implies a compression of the o-rings of 0.005", as expected.

AI #15:
Write a procedure for installation of Collar Assembly onto the AC flat.

ACTION ITEM IS CLOSED: 9/14/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

The collar assembly installation procedure DCATT-PROC-001 was written 1/26/98. It will be updated and reissued to reflect the current situation with polishing done at Zygo.

03/22/99

AI #16:

Write a procedure for receiving AC flat from Zygo, installing collar assembly, rotating, moving into the CIA room. Include an inspection for new dimensions of the AC flat and the shipping method from Zygo to GSFC. [Hakun, Moya]

ACTION ITEM IS CLOSED: 10/2/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

The procedure is written in DCATT-MECH-PROC-002. The new dimensions of the AC Flat are 40.070 inches diameter, 7.372 inches thick. 0.028 inches was removed from the thickness during the grinding and polishing process, per Zygo.

AI #17

Do an analysis on what happens if the AC Flat Support Frame is tipped at maximum amount on jack screws. Will there be a safety hazard? [Moya, Hakun]

ACTION ITEM IS CLOSED: 12/4/98

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

The machine screw jacks have a maximum rated travel of 6 inches. At about 8 inches of travel, the lead screw disengages from the brass nut and is no longer safe at any load. For DCATT the lifting height will be set at 0.50 inches from the bottom position. The tip and tilt range requirement for the AC Flat is 4 arc minutes, the maximum Z-axis travel of the jack is only 0.064 inches. Thus there is no need to allow the jack screw to travel more than about a half-inch in either direction. A limit switch, or arresting cable, in addition to software controls, will be installed on the jack screw to prevent a total lift height of more than 1.0 inches. This will limit the tilt to a maximum of 1.05 degrees. Thus the side load into the three jack screws and Telescope Support Frame legs will be:

$W*\sin(1.05) = 25$ pounds force,
when W = mass of AC Flat Support Frame = 1353 pounds

The 25 pounds of force will be divided among 3 legs.

As part of action item #13, the jack screws were tested for side load. The Duff-Norton MS-2501 machine screw jack is rated to lift 2000 pounds vertically. Its actual failure point is at 4000 pounds, according to data supplied by the vendor. Its horizontal load capability (side thrust rating) is 89 pounds for a 400 pound compression load with a 3" raise. The three DCATT jack screws lift 1352 pounds less than 1" above their lowest position. When they are all at the same height, the load is spread evenly at 1352/3 = 451 pounds apiece.

The jack screws were tested on 12/3/98 to verify whether they would see a side load that could cause a safety hazard. In the actual AC Flat Support Frame, the jacks were raised 3.75 inches one at a time. In one configuration, two jacks were up 3.75 inches and the other was at its lowest position. In another configuration, one jack was up 3.75 inches and the other two were at their lowest position. In neither of these situations was there any binding, breaking, or other anomalies noted. In this latter configuration, a tilt of 2.88 degrees exists. The side load exerted in this case is 68 pounds of lateral force distributed unevenly on the 3 jack screws. The test results showed that the jack screws are safe when the maximum difference between jacks is less than 3.75 inches. There is a conservative distance since the jacks were not tested to failure.

AI #18

Check ball bearing plate to steel leveling pad interface static friction coefficient. If too high, then select more compatible materials [Davila]

ACTION ITEM IS CLOSED: 10/2/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

This action item is concerned with the interface between the AC Flat Support Frame leveling pad and the "dog bowl" on top of the machine screw jack. Does the interface allow the leveling pad to slide in the dog bowl when the frame is tipped or tilted? If not, then tip/tilt resolution could be affected, since the frame is not acting as a rigid body. When the AC Flat Support Frame is tilted, the distance between the three legs in the X and Y axes is changed. The Frame can not be tilted more than 1.05 degrees, or 1.0 inches (see AI #17) in the Z-axis which causes a maximum of 0.011 inches travel in X and Y axes of a leg. Moments would be induced into the legs of the AC Flat Support Frame resisting the tip or tilt. These moments in turn induce strain in the legs which introduce errors to the position of the AC Flat on the order of 64 nanometers in the Z-axis, assuming the Frame acts as a rigid non-deformable body.

Realistically, if there was no movement in the leveling pad (assume it was bolted in place), a stress of 4500 psi would be induced into the leg of the Frame. [note: 4500 psi is within acceptable limits of strength]. This stress would pull the leg longer by the required amount. In effect, the Frame would act like a flexure or a spring.

For DCATT, the interface is a commercially available "ball bearing plate" consisting of a _" thick steel plate with 5 stainless steel ball bearings embedded into it.

The static coefficient of friction between the polished stainless steel ball bearings and the 303 stainless steel cup is approximately 1.8 for a sliding non-lubricated contact [Source: *Vukabrotavich GSFC Short Course, 1998*]. This number is not relevant, however, since the rolling friction is what governs the movement of the bearings. Rolling friction is "much less than sliding friction" [*Physics*, Resnick & Halliday, p.100], thus the friction is low enough that the balls will roll, the leveling pad will move, and moments induced into the Frame will be low, and errors on AC Flat position will be less than 64 nanometers in the Z-axis.

AI # 19
Recommend closing off tubes on Upper T-frame to prevent contamination of optics later on from dust buildup. [Wallace]

ACTION ITEM IS CLOSED: 9/14/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

David Robinson issued Revision A to drawing 2022803 calling for tubes to be closed off. Drawing sent to Bechdon (fabricator) 8/21/98.

AI #20

Consider stress relieving secondary mirror.

ACTION ITEM IS CLOSED: 10/28/98

Responsible Engr: Jamie Britt

Project Manager: Claudia LeBoeuf

The secondary mirror will be stress-relieved using the same heat-treating steps that were used on the primary mirror segments. The mirror will be rough machined, inspected, and then annealed according to the procedure laid out in Section 3.1 of DCATT PROC-001, Fabrication Procedure for DCATT Segmented Primary Aluminum Mirror. A second inspection will be done after annealing. Then the mirror surface will be diamond turned. No further stress relief will be performed on the segment.

AI # 21

Determine the stress on the AC flat during 2 point rotation. Do a Weibull analysis [Hakun]

ACTION ITEM IS CLOSED: 10/9/98

Responsible Engr: David Robinson, Jamie Britt

Project Manager: Claudia LeBoeuf

When the AC Flat is rotated 180 degrees from shiny side up to shiny side down, it will briefly be in contact with two 1" diameter leveling pads. The leveling pad surface is Buna-N rubber. A stress analysis of the AC Flat during the 2-point rotation shows that the maximum stress is 540 psi in compression against the side of the glass. This was determined by dividing the area of contact of the two 1.0" diameter levelling pads by the weight of the AC Flat. The compressive strength of Zerodur, as well as all glasses, is 1-2 orders of magnitude higher than its tensile stress. A number of 50,000 psi is used by Vukabrotovich. Thus, with a factor of safety of 5.0, the margin of safety is 18.5.

This situation will only occur for <10 seconds during rotation. Two-point rotation is only expected to occur once.

Weibull Stress Analysis of 2-Point Rotation of AC Flat

Vukabrotovich recommends that a Weibull analysis be done on glass which is to be used in tension, such as windows in a pressurized container. Because glass is brittle, it fails by cracking. The stress which cracks the glass is related to initial flaw size, and flaw sizes are a statistical function of material, surface roughness, and the fabrication process. Given that no glass is flawless, there is always a small chance that glass can fail, no matter what stress it sees. The goal here is to show that the probability is very low. A Weibull statistical analysis has been performed for the case of a 2-point rotation of the AC flat in its collar to determine if the mirror will be damaged as it passes through 90 degrees of rotation.

Assumptions: For this analysis, the flat was assumed to be made of Zerodur. No scaling was made for crack size. This means it is assumed that cracks in the mirror are not larger than cracks in the test pieces from which the Zerodur Weibull parameters were obtained. The stressed area was assumed to be equal to the area of the 1-inch diameter pads.

Cases: Four cases were examined. Case 1 assumes a flaw depth of 298 μm. Case 2 assumes that the mirror flaws correspond to those in an optically polished test piece. This turns out to be a worse case than Case 1, although it may not be accurate since the mirror surface that contacts the pads is not polished. Both cases were also examined using an arbitrarily chosen larger stressed area. These are Cases 3 and 4.

Weibull Results

Stressed area = 0.001 m^2 (area of two pads)
Case 1: chance of failure = 1 in 1.387 million
Case 2: chance of failure = 1 in 94,000

Stressed area = 0.298 m^2 (half the side area of the mirror: overly conservative)
Case 3: chance of failure = 1 in 4,700
Case 4: chance of failure = 1 in 320

Conclusion:

Depending on the assumptions made, the probability of failure ranges from .003 to
.0000007.

Given that the compression stress margin of safety is quite high and that the Weibull
probability of failure is quite low, the AC Flat is expected to survive the rotation
procedure.

AI #22

Verify safe condition of AC flat during shipment from Zygo, given that it will no longer have same thickness as before. Does the collar still work? [Hakun, Moya, Wallace]

ACTION ITEM IS CLOSED: 10/2/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

The AC Flat thickness was 7.400" before polishing and 7.372 inches after [per Zygo measurement]. The Ring Segment piece (drawing # 2022777) is 7.50 inches thick. The lip segment o-ring groove is 0.18" deep and the o-ring is 0.25" in diameter. Thus the O-ring sticks out from the groove 0.070 inches. When compressed by the 7.400 inch thick AC Flat, the compression on the O-rings was 0.005 inches (see AI #14). Thus the O-ring stuck out 0.065 inches from the groove under compression. This means that the lip segment bends or yields slightly to accommodate the extra 0.015 inches that the O-ring sticks out.

$$0.065 - (7.500 - 7.400)/2 = 0.015"$$
[note: divide by two since o-rings on top and on bottom of ring segment]

This is desirable and the interface was intentionally designed that way.

At the AC Flat's current thickness of 7.372 inches, the o-ring will stick out 0.001 inches.
$$0.065 - (7.500 - 7.372)/2 = 0.001"$$

Thus there will still be compression on the newly polished AC Flat, and the Collar Assembly is still acceptable to use.

AI #23

Analyze the self-weight gravity sag of secondary mirror design

ACTION ITEM IS CLOSED

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

The self-weight gravity sag of the secondary mirror is calculated to be ***0.03 waves*** peak to valley. The analysis is documented on the website:
http://cormorant.gsfc.nasa.gov/~mark/DCATT/secondary/

The text of the memo is as follows:

TO: 544/Jamie Britt
 FROM: 542/Mark McGinnis
 SUBJECT: Secondary Mirror Mount Analysis

Code 542 has completed study of the gravity sag of the DCATT secondary mirror when inverted relative to it orientation during polishing. The error is well within the budgeted amount.

The DCATT secondary mirror studied is a 7.874 inch diameter cylindrical optic with a height at the perimeter of 1.935 inches. It is made of 6061 Aluminum. It mounts at three points separated by 120 degrees using 6 mm screws passing through a .5 inch thick flange. The .75 inch long stem region has an outer diameter of 6 inches. It is light weighted by boring a 4 x 1.75 inch cylinder in from the mounting surface.

The mirror was modeled using NASTRAN solid elements. A 120 degree segment of the mirror was modeled to reduce computer run time. It was constrained in translation at three points to conservatively represent the attaching screws. It is shown in Figure 1.

The applied load is 2 g's normal to the mirror surface. This models the gravity induced deformation since the mirror is inverted from its orientation during polishing. The peak-to-valley error is 0.83 micro-inches (.03 waves). It is shown in Figure 2. The budgeted peak-to-valley error is 1/10 wave. The expected value is 30% of the budgeted value.

AI #24

Determine worst-case shock loads and results on DCATT caused by isolator failure, either abrupt loss of pressure or wild swing back and forth.

ACTION ITEM IS CLOSED

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

Newport was contacted about the maximum travel possible on the isolators and the maximum velocity that they could move. The maximum travel of an isolator is _" in piston. In a worst-case scenario, the DCATT assembly could tip _" over its 120 inch width. This produces a tilt of 0.36 degrees. The Newport chief engineer stated that the isolators have benign failure modes. For example, when an isolator is floating, there is no failure mode where the air suddenly leaves the isolator, and the floated item free-falls to a hard stop. The isolators have an airflow restrictor to slow the air as it leaves the chamber, so that the result of some kind of pressure leak is one of slow sinking to a hard stop.

Wild oscillation is possible if the isolators are too highly pressurized and are not balanced properly. The control for this is to start the system at a low pressure (10-20 psi) and gradually increase it until the system is floating. Without the AC Flat Support Frame, this point is reached at about 45 psi. Also, as has been recognized early on, there is an envelope for the center of gravity of the system. If the CG is outside of the envelope, the isolators can become unstable. DCATT has been designed to avoid this problem.

An analysis was performed by Mark McGinnis/542 to determine the loads and stresses involved in a failure of the isolators. This memo is located at http://cormorant.gsfc.nasa.gov/~mark/DCATT/visf/

TO: 543/David Robinson
 FROM: 542/Mark McGinnis
 SUBJECT: Failure of the DCATT Vibration Isolation System

Code 542 has studied the likelihood and effect of catastrophic failure of 2 of the DCATT vibration isolation system airbags resulting in a free-fall of the DCATT structure onto the hard stops of the isolators. Failure of a single isolator pair would not result in a free fall condition as four actuators are capable of supporting the structure's weight. The structure is found to be capable of surviving a 3/4 inch fall.

The best method of determining the effect of a free fall is through conservation of energy methods. The kinetic energy of an 8000 lb. table falling .75 inches is 6000

in.-lb. The structure will not fail if this amount of energy can be absorbed by the structure through elastical deformation. It was assumed for this analysis that the energy absorbtion would be through bending in the three 8 inch box beams beneath the optical bench. They are 107 inches long with an inertia of 77.7 inches4. To simplify the problem each beam was analyzed as a 53.5 inch beam clamped at one end and perpendicularly loaded by a force of magnitude P at the free end.

The strain energy (U) for a beam in bending is $U=P2L3/6EI$. To avoid failure each of the 6 half beams must absorb 1000 inch-pounds. Solving for the value of P which expends 1000 inch-pounds yields:

P=(6 E I U/L3)0.5
P= (6 * 30E6psi * 77.7in4 * 1000in-lb/(53.5in)3)0.5
P= 9560 lb.

The stress resulting from a 9560 lb perpendicular load is

f=Mc/I =Fdc/I
f=9560lb (53.5in) (4in)/ 77.7in4
f=26.3 ksi

The yield stress for the A-500 steel in these elements is 46 ksi. A factor of safety of 1.0 is used since this already assumes to failures. The resulting M.S. is 0.75.

The DCATT structure has been found to be capable of surviving the unlikely event of sudden failure of two support structures.

AI #25

Determine modes of entire DCATT assembly and its response to the acceleration environment in clean room. Determine motion of AC Flat in hindle mounts. Determine motion of primary and secondary mirrors. Determine if significant loads are induced into epoxy joints or jack screws.

ACTION ITEM IS CLOSED

Responsible Engr:

Project Manager:

The response to this action item is complex. The last sentence can be dismissed, however, by understanding that the vibration environment in the room is on the order of millionths of a g. Thus no significant dynamic loads will be introduced into the epoxy joints or jack screws.

Presently (1/20/99), DCATT is fully modeled in NASTRAN. However, finding the natural frequencies of the various components does very little good unless the forcing function of those parts is understood. For example, if a part has an alarmingly low natural frequency (like 3 Hz) but no forcing function exists in that frequency, then no effect will be seen. Likewise, a component with a high natural frequency (like 120 Hz), might be unluckily tuned to a forcing function in the room and it would vibrate significantly. Lastly, according to John Hagopian, DCATT's alignment engineer, there is no way to predict whether interferometry may be taken in a given setup, because the Zygo has unpredictable modes, and interactions between optics may or may not be constructive. The simple answer is that DCATT must be tested optically.

Nonetheless, a very serious effort has been underway since 10/98 to measure the vibration environment in the CIA room. Micro-g accelerometers were purchased and have been used to measure the vibration spectrum on the AO Bench and on the floor. Data was taken with the AO Bench floating and not floating. NASTRAN analyses used this data to determine the relative displacement of DCATT optics. The resulting motion of the optics after this data was taken through the analytical models was larger than had been hoped for, but also seemed unrealistic when compared with the fact that good optical measurements, including interferometry, are being taken with success on the AO bench. The previous assumption was that the vibrations were coming from the floor, going through the isolators and up through the structure. Now, however, the project has come to the conclusion that a large portion of the vibrations measured by the micro-g accelerometers was actually coming from wind and acoustics in the room. The forcing function is therefore applied randomly all over the structure. Thus all NASTRAN analyses to date are invalidated, since the underlying assumption was incorrect.

Vibration data is presently being taken in the CIA room with the fans off, the AO Bench floating in the pneumatic isolators, and the AO Bench Frame entirely on the seismic

block in the CIA room. This should show us the best environment we could hope to see in there.

To attenuate wind and acoustic vibrations, some sort of shroud or tent may be built around DCATT. It is the opinion of Bill Eichhorn and the team that this is a step by step process of isolating problems through primarily optical measurements and then dealing with them specifically. As he pointed out, if we were to build a shroud right now to deflect room air, it might cause worse problems with thermal currents. The project will continue the process of defining what it requires. The full-up structure, including telescope structure, will be integrated in February 1999 and will have dummy masses for all optical elements not yet in-house. We will then conduct a detailed characterization optically and mechanically of the vibration environment.

AI #26:

To provide more lateral stiffness, reduce the extended length of the 1-Ton machine screw jack. Set nominal position at +5.5 inch/-.5 inch, not +/- 3 inches travel, for example. [Ryan]

ACTION ITEM IS CLOSED: 9/14/98

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

Agree. The maximum Z-axis pistoning expected is 4 arc minutes tip/tilt. This corresponds to 0.1" maximum Z-axis pistoning. Thus the machine screw will be biased so that it is only 0.5" above its bottom position.

AI #27:

Recommend that all welded structures are stress relieved via heat treatment to minimize concerns of creep. [Hakun, Moya, Wallace]

ACTION ITEM IS CLOSED: 10/9/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

The items of concern are the AO Bench Frame, the Telescope Support Frame, and the AC Flat Support Frame.

The latter two structures are made from stress relieved 6061-T6511 tubing so the tubing is not expected to creep. The welded joints are the only concern. According to the Metals Handbook, the heat-effected zone of weldments have become annealed, thus stress relieved by the very action of welding. However, many metals handbooks state that post-welding stress relieving is desirable if extreme stability is required.

The steel AO Bench Frame was made from ASTM A500 steel tubing, not stress relieved. It is so large that the GSFC Planning Office does not know of any facility big enough to handle a post-weld stress relief operation. It might require placing it in some steel mill's foundry furnace. This is not considered feasible since it would have significant cost and schedule impacts on DCATT.

There are three factors that control creep: temperature, stress, and time. The temperature of the welded components will be controlled in the cleanroom to 65 +/- 5 degrees F. These weldments will not be subjected to cryogenic temperatures. The Metals Handbook suggests that if metals are not subjected to temperatures greater than half their melting temperature, creep is not significant. The Handbook also stated that if the stress on parts is below 20% of their yield strength, then creep would not be significant. The highest stress of any non-stress relieved weldment is as follows:

Weldment	Max Weld Stress	Yield Strength	% of Yield
AO Bench Frame	2000 psi	46000 psi	4.3%
Telescope Support Frame	1000 psi	33000 psi	3%
AC Flat Support Frame	<200 psi	33000 psi	<1%

The last factor is time. All metals will creep with time at an atomic level and millenium time scales. DCATT's components must remain stable on a time constant of 24 hours.

After extensive research at the GSFC library, no useful data or analysis methods could be found on microcreep. Several experienced engineers including Armando Morell/544 and Lee Niemeyer/543, and Mike Wade/547 believe that there will not be significant creep in the weldments. Given the relatively small time period, low stress, and constant low temperature, the creep should be negligible for DCATT's purposes. Characterization tests, including optical measurements will be taken in October and November to verify that there will be no problem with creep. If a problem is encountered, then the problem will be addressed further.

AI #28

Recommend that Materials Group "bless" design of epoxy joint and the epoxy chosen. [Moya, Hakun]

ACTION ITEM IS CLOSED: 12/4/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

Several Goddard experts have been consulted, including Michael Viens and Carol Clatterbuck of the Materials Branch, Jim Lyons and Bill Eichorn of the Optics Branch, Armando Morell and Israel Moya of the Electromechanical Branch.

Carol Clatterbuck stated that the most important parameter for the bond is the preparation of the surfaces. The invar mushroom surface should be sanded with 100 grit sandpaper. The AC Flat surface can be used as is, or could be roughened to 220 grit roughness to match the earlier tensile test condition. Then, both surfaces should be cleaned with acetone using extracted cotton wipes. The area should be thoroughly cleaned several times. Then a final cleaning using ethyl alcohol is advised. The invar mushrooms then should be dried in an oven (in air) at 65 degrees C for one hour. The AC Flat is impractical to dry this way so it should be dried in air for at least one hour. Both the AC Flat and the mushrooms should be ready for the epoxy at the same time. The bonding should take place on the same day as the cleaning to minimize dust accumulation, etc.

The epoxy is 3M's 2216 Clear. Two pull tests were performed on Zerodur substrates with invar mushrooms. The bonds failed in adhesion at 1688 and 1396 psi of tension, as documented by Michael Viens memo of 4/8/98.

There are nine mushrooms epoxied to the back of the AC Flat. The AC Flat weighs 850 pounds. Thus each mushroom will see about 95 pounds. Each mushroom's bonded surface area is 1 square inch, thus the tensile stress on the epoxy is 95 psi. This provides a margin of safety of at least 10 on the bond.

The bond will be tested before it is placed 15 feet up in the DCATT assembly. After proper curing, the AC Flat will be lifted using the Hindle Mounts, so that it is suspended no more than _ inch above the lower lips of the collar assembly. Thus if there is a failure, the flat can fall no more than _ inch into the collar assembly. The flat will remain suspended for at least 1 month to test the long term stability of the bonds.

AI #29

Recommend revisiting design of Hindle mount mushrooms and use of spherical washers. Explain the function that the spherical washers provide.

ACTION ITEM IS CLOSED: 9/14/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

Two spherical washer sets are used on the Hindle Mount Assembly (drawing 2022788A) to provide degrees of freedom in rotation about X and Y. One spherical washer set is located on top of the triangular plate and the other is located below the triangular plate.

The reason for these washers is to avoid placing the epoxy joint of the Invar mushrooms to AC Flat in shear. When the epoxy joint is created, the mushroom may be very slightly non-parallel to the mirror surface. Also, the mushroom top and bottom surfaces may be slightly out of parallel, on the order of 0.001 inch or less. These non-parallel surfaces cause slight cocking of the mushroom relative to the plane of the mirror surface. When the bolt is fastened, a tangential shear load will be induced into the epoxy joint on the order of the sine of the angle of cocking. This load will be small since the angle is small. However, this load is eliminated with the spherical washer set located between the mushroom and the triangular plate.

If the bolt-hole in the Invar mushroom is out of perpendicularity with the bottom surface of the mushroom, the bolt head would be cocked relative to the plane of the triangular plate, and the clamping force would not be uniform. The spherical washer set between the bolt head and the triangular plate serves to make the clamping force between the bolt head and the triangular plate uniform.

If the bolt-hole is not perpendicular to the bottom surface, a shear force is created [again, sine of the angle] in the epoxy joint when the bolt is torqued, regardless of the use of the spherical washer. However, the method of drilling and tapping holes in the mushrooms using a numerically controlled milling machine ensures perpendicularity within arc seconds.

AI #30

What is resolution requirement on tip/tilt of AC flat?

ACTION ITEM IS CLOSED: **9/14/98**

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

The resolution on tip and tilt of the AC Flat is 1 arc second.   The total range of motion is 4 arc minutes. This requirement is documented in DCATT-DOC-001.

AI #31

Machine screws may drip oil and contaminate mirror. Recommend bagging screws or building a "well" around screws. The well may be advisable for safety reasons anyway, to ensure if jack screws tipped over during earthquake.

ACTION ITEM IS CLOSED: **9/14/98**

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

Agree. Either a sheet metal "bucket" will be installed around the machine screw jack or the jack will be encased in cleanroom compatible plastic, such that no grease could drip out and contaminate the primary mirror.

Neither solution will provide additional safety if the jack screws "tip over". This hazard will be controlled by proper mounting of the jack screw to the frame, using it at one-fourth of its rated load, using it at nearly its fully retracted position to reduce the moment arm, and proving by analysis that a worst-case failure scenario of runaway pneumatic isolators causing rapid pistoning of the frame side to side will not cause catastrophic failure or tipping of the screws (see AI # 17).

AI #32

What is the strength of the 1/4-20 leveling pads?

ACTION ITEM IS CLOSED: 10/9/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

According to the vendor of the leveling pads (Reid Tool Supply Company), the _-20 black chromate steel leveling pads (part #LBK-6) have a rated compression load of 1080 pounds.

The bolt of the leveling pad will never be placed in tension or shear.

AI # 33

Recommend measuring AO Bench Frame before it goes into the clean room. Understand its out-of-straightness condition before mounting AO Bench. [Niemeyer, et al]

ACTION ITEM IS CLOSED: 9/14/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

This is a good idea, however, the AO Bench Frame was installed in the cleanroom before this could be implemented.

Coplanar errors in the AO Bench Frame do not affect the optical system except if the Telescope is cocked relative to the AO Bench. This cocking could occur if the three support frame pads are not coplanar. The frame was inspected at the vendor and found to be within the tolerances specified on the drawing 2022800A. This means that the three pads are coplanar with the 6 pneumatic isolator interface planes within 0.1".

Perpendicularity of the Telescope relative to the AO Bench will be assured by use of a kinematic interface ring between the Telescope and the Telescope Support Frame that will compensate for any non-perpendicularity in the frame itself.

AI #34

Go over epoxy procedure with Israel Moya.

ACTION ITEM IS CLOSED: 12/15/98

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

This was done 12/15/98. Mr. Moya was in agreement with the proposed plan of action. This plan will be written in DCATT-MECH-PROC-006, the epoxy procedure.

AI #35

Recommend getting Quality Assurance involved on the epoxy bond. [Wallace]

ACTION ITEM IS CLOSED: 12/4/98

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

Since DCATT is not a flight project, quality control on the epoxy procedure will be performed by the DCATT lead mechanical engineer.   The epoxy procedure will be monitored at all times. Any deviations from the signed and approved epoxy procedure will require project approval. The work will be done by personnel experienced in the use of these adhesives.

AI #36

Recommend writing up a test plan and executing test for finding the backdrive load on machine screws, the horizontal load capability, and how safety will be assured during operation of machine screws if driven to full tip/tilt [Wallace]

ACTION ITEM IS CLOSED: 10/9/98

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

As detailed in AI #12, a test of the backdrive of the Duff-Norton 1 ton machine screw jack was performed. The backdrive was zero, at loads up to 1500 pounds in compression. Since the maximum load on a jack is about 500 pounds, backdriving will not occur.

The AC Flat Support Frame sits on three leveling pads which have up to 15 degrees of tilt capability. The leveling pad rests on a ball bearing plate. This plate is captured inside a stainless steel cup. The cup has a threaded fitting on its bottom surface at the centerpoint. When the AC Flat is tilted due to pistoning of the jacks, the leveling pads will rotate to keep themselves flat and parallel to the cup. As the Frame is tilted out of its true flat position, the compression load (which was evenly distributed) on each jack changes according to the sine of the angle. Since the Frame is free to move in X and Y because of the ball bearing plate interface, there are no side loads exerted on the jack. Side loads are not an issue.

According to the Duff-Norton jack handbook, the horizontal load capability is 89 pounds for a 400 pound lift of 1.8 inches. Duff-Norton did not provide details about possible factors of safety on the side load.

Jack Screw Side-Load Rating

The machine screw jacks will be used at the lower end of their range of motion, never exceeding 0.6" from bottom dead center. However, a worst-case of 6" could be possible but highly unlikely. The gear ratio is 100:1, so it would require someone to turn the input shaft 600 times – a very tedious task which would take over 10 minutes. Assuming this happened, if one jack is raised 6" and the others are lowered to 0, the AC Flat Support Frame would tilt 6.25 degrees in one orientation and 4.3 degrees in the other.

The jack's failure mode is yielding of the drive nut inside the machine screw jack. The drive mechanism is within a steel housing. Yielding the drive nut yields the drive mechanism, but does not break the piston shaft, so the load path would still be maintained. This is a graceful failure, because failure of the drive mechanism halts further tipping of the AC Flat Support Frame. Of course, the jack would have to be replaced.

NOTE: ADDITIONAL INFORMATION AFTER ACTION ITEM WAS CLOSED:

The jack screws were tested on 12/3/98 to verify whether they would see a side load that could cause a safety hazard. In the actual AC Flat Support Frame, the jacks were raised 3.75 inches one at a time. In one configuration, two jacks were up 3.75 inches and the other was at its lowest position. In another configuration, one jack was up 3.75 inches and the other two were at their lowest position. In neither of these situations was there any binding, breaking, or other anomalies noted.   In this latter configuration, a tilt of 2.88 degrees exists, and 68 pounds of lateral force distributed unevenly on the 3 jack screws. The test results showed that the jack screws are safe when the maximum difference between jacks is less than 3.75 inches. There is a conservative distance since the jacks were not tested to failure.

AI #37

Recommend butt milling support beams [Wallace]

ACTION ITEM IS CLOSED

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

This action item was part of a list of improvements on the design of the entire DCATT assembly, and should be considered if DCATT requires more large-scale support frames.

The AO Bench Frame was delivered to GSFC on August 17. The fabrication of the other support frames has also been completed. It is not feasible to incorporate this recommendation in the current design for cost and schedule reasons. The DCATT assembly will be tested in the CIA room in October and November, 1998. It is fully expected that these tests will show that the design of the large frames is good enough and meet requirements.

AI #38

Recommend checking temperature of CIA room at various sites and elevations [Wallace]

ACTION ITEM IS CLOSED: 10/9/98

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

Agree. This will be done as part of the DCATT characterization testing per test plan DCATT-TP-001.

AI #39

Recommend measuring flatness of mounting pads and AO Bench frame. [Wallace]

ACTION ITEM IS CLOSED: 9/14/98

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

This is a good idea, however, the AO Bench Frame was installed in the cleanroom before this could be implemented. The frame was inspected at the vendor and found to be within the tolerances specified on the drawing 2022800A. This means that the three pads are coplanar with the 6 pneumatic isolator interface planes within 0.1".

Coplanar errors in the AO Bench Frame do not affect the optical system except if the Telescope is cocked relative to the AO Bench. This cocking could occur if the three support frame pads are not coplanar.

Perpendicularity of the Telescope relative to the AO Bench will be assured by use of a kinematic interface ring between the Telescope and the Telescope Support Frame that will compensate for any non-perpendicularity in the frame itself.

AI #40

Write a briefing paper that describes the various metering structures considered and why present design was chosen. Include also preliminary design of Serrurier truss metering structure. Include cost/schedule impacts for going with Serrurier truss. Include also results of metering structure characterization tests.

ACTION ITEM IS CLOSED

Responsible Engr: David Robinson

Project Manager: Claudia LeBoeuf

The briefing paper was written and numbered DCATT-MECH-DESI-002.

AI #41:

Consider hard stops under AO Bench Frame in case of failure mode on isolators that could cause entire tower stackup to sway uncontrollably.

ACTION ITEM IS CLOSED: 10/9/98

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

Hard stops under the AO Bench Frame would serve to limit the range of motion of the AO Bench Frame in case of a failure mode where the isolators pistoned uncontrollably. The isolators have a total piston range of _". The entire AO Bench Frame, if one end was _" higher than the other, would be tilted by 0.3 degrees.

These hard stops may in fact cause instability by limiting the ability of the isolators to damp out large amplitude disturbances. It is the opinion of Lou Worrell, DCATT's pneumatic isolator technician, that this may do more harm than good.

A call was made to Newport's pneumatic isolator engineer, Mr. Bowie Houton. He believed that hard stops might be useful to limit the range of motion of the isolators; however, he felt that since the entire range of motion was only _" of an inch, the risk of anything pitching over, tipping, or moving around was extremely small. Also, Mr. Houton that design improvements in the last 10 years have reduced the risk of uncontrolled oscillation, given that the CG is properly located and the isolators are not excited (with huge forces) at their natural frequency of 1.2 hertz.

The side load exerted by this tiny tilt is found by multiplying the mass of the structure by the sine of 0.3 degrees. The entire mass of DCATT (6000 pounds) times the sine of 0.3 degrees equals 28 pounds.

Given that the risk of uncontrollable oscillation is very small, the range of motion is small, and the side loads exerted are small, blocks are not warranted. Characterization testing in the CIA room with the DCATT structures will verify this conclusion.

AI #42

Write hexapod prototype test plan

ACTION ITEM IS CLOSED

Responsible Engr: Jamie Britt

Project Manager: Claudia LeBoeuf

This test plan is documented in DCATT-MECH-PLAN-005.

AI #43

What is mass of lightweighted segment compared to un-pocketed segment?

ACTION ITEM IS CLOSED: 12/4/98

Responsible Engr: <u>David Robinson</u>

Project Manager: <u>Claudia LeBoeuf</u>

The mass of a lightweighted mirror segment (outside segment, not center segment) is 6.393 pounds. The mass of a mirror blank, with rough machined figure but no lightweighting pockets, is 14.204 pounds.

Thus the ratio is 6.393/14.204 = 0.45

Looked at another way, the lightweighted segment is 55% lighter than a solid segment.

AI #44

Consider a modal survey of hexapod assembly to verify model

ACTION ITEM IS CLOSED: 2/1/99

Responsible Engr: Jamie Britt

Project Manager: Claudia LeBoeuf

A modal survey was performed on the prototype in mid January . The data correlated very well with the NASTRAN model. Results are posted at
http://cormorant.gsfc.nasa.gov/~mark/DCATT/hex_dyn/

AI # 45

Can an actuator be removed and replaced if it is broken?

ACTION ITEM IS CLOSED: 9/17/98

Responsible Engr: Jamie Britt

Project Manager: Claudia LeBoeuf

An actuator can be removed and replaced if broken, although the procedure will not be trivial. Replacing an actuator would require removal of the mirror segment from the hexapod and partial disassembly of the hexapod. It is unlikely that this would be done with the telescope on the testbed, although it might be possible. It is more likely that the telescope would be removed from the testbed, requiring removal of the AC flat as well, and placed on the primary mirror integration stand. Some bracing of the metering structure might need to be removed during this operation to provide better access to the segments.

ADDITIONAL INFORMATION AFTER ACTION ITEM WAS CLOSED:

A new primary mirror baseplate is being designed. This baseplate will not have the holes underneath the mirror segments and the hexapod bucket concept. It will be a simple 2" thick plate that the hexapods will sit upon. Thus it will be easy to remove and replace any of the hexapod assemblies without disturbing the others. This new baseplate should be designed by February 1999.